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TECHNICAL NOTE

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MEASUREMENTS OF RECOVERY FACTORS AND COEFFICIENTS OF HEAT TRANSFER IN A TUBE FOR SUBSONIC FLOW OF AIR

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SUMMARY

Measurements of heat flow to air at subsonic velocities and at substantially constant Reynolds number show that the heat-transfer coefficient h_e, based on the difference between the temperature of the heated wall and the adiabatic wall temperature, is independent of this difference. In order to determine the adiabatic wall temperature, recovery factors were measured at the pipe wall for adiabatic flow. The recovery factor averages 0.88 and is substantially independent of Mach number in the range from 0.2 to 1. The coefficient of heat transfer h_s, based on the difference between the temperature of the heated wall and the mean stagnation temperature of the stream, is not independent of this temperature difference unless the temperature difference is large compared with the difference between stagnation temperature and mean stream temperature. The conventional heat-transfer coefficient h_m varies even more with temperature difference. The preferred Stanton number h_e/c_pG, where c_p is specific heat at constant pressure and G is mass velocity, is nearly independent of average Mach number in the range from 0.1 to 0.75, and varies with Reynolds number substantially in the manner characteristic of turbulent flow of incompressible fluids in pipes.

Published data for flow of air at high Mach numbers involve such large temperature differences that they throw no light on whether h_e or h_s should be employed for heat-transfer calculations. They are used here to extend the

present conclusions to much higher Reynolds numbers and temperature differences, leading to the relation

$$\frac{h_e}{c_p G} = 0.033 \left(\frac{DG}{\mu_m}\right)^{-0.23}$$

where D is the inside diameter and μ_m is the absolute viscosity of air at average mean stream temperatures.

INTRODUCTION

In heat transfer to or from a stream of incompressible fluid the coefficient of heat transfer is defined as the rate of heat transfer per unit of surface area per unit difference between the temperature of the surface and the mean stream temperature at the cross section in question. This "mean coefficient" of heat transfer is expressed as

$$h_m = \frac{dq}{dA} \frac{1}{(t_w - t_m)} \tag{1}$$

(All symbols are defined in the appendix.) The value of this coefficient is found to be substantially independent of the temperature difference except for large temperature differences.

In heat transfer to or from a stream flowing at high velocity the value of this same coefficient is no longer independent of the temperature difference, particularly for small temperature differences. For example, when heat transfer to or from the surface is zero (i.e., for adiabatic flow), the temperature of the surface may be greatly in excess of the mean temperature of the adjacent stream, and the coefficient in question is therefore zero. If the temperature of the wall is increased so as to cause heat transfer to the fluid, the coefficient becomes greater than zero. If the temperature of the wall is decreased somewhat so as to cause heat transfer from the fluid, the coefficient becomes less than zero. A coefficient with such characteristics is of no utility.

For compressible fluids the coefficient of heat transfer may be redefined in terms of the difference between the surface temperature during heat transfer and the surface temperature in the absence of heat transfer to insure that the temperature difference vanishes with the heat transfer and to preclude negative values of the coefficient. The surface temperature in the absence of heat transfer is referred to throughout this report as the "adiabatic wall temperature." The corresponding coefficient of heat transfer is termed the "effective coefficient of heat transfer" and is written

$$h_e = \frac{dq}{dA} \frac{1}{(t_w - t_{aw})} \tag{2}$$

An alternative definition for the coefficient of heat transfer may be given in terms of the difference between the surface and the stagnation temperatures of the stream, where the stagnation temperature is the temperature that the stream would have if it were mixed and its velocity were adiabatically reduced to zero. This definition is at best an approximation of the previous one. The "stagnation" coefficient of heat transfer would be written

$$h_s = \frac{dq}{dA} \frac{1}{(t_w - t_s)} \tag{3}$$

Unless the stagnation temperature and the adiabatic wall temperature are identical, this coefficient, like that based on the mean stream temperature, may be greater than, equal to, or less than zero.

The "recovery factor" r is defined as the ratio of the excess of the adiabatic wall temperature over the mean stream temperature to the excess of the stagnation temperature over the mean stream temperature. For a gas having the equation of state

$$pv = RT$$

the stagnation temperature is the same whether the reduction to zero velocity occurs at constant pressure (as through friction) or reversibly (as in diffusion). The recovery factor is given, therefore, by the expression

$$r = \frac{T_{aw} - T_m}{T_s - T_m} = \frac{t_{aw} - t_m}{V_m^2/2g_c J c_p} \tag{4}$$

Although recovery factors have been measured for adiabatic flow of air past plates (references 1 and 2), parallel to wires (references 3, 4, and 5), and normal to single cylinders (references 4 to 8), no published data are available for recovery factors for flow of gas inside cylindrical tubes. In a paper by Kalitinsky and Hottel presented before the annual meeting of the American Society of Mechanical Engineers in December 1943, measured recovery factors were reported for adiabatic flow of air past plates.

References 9 and 10 give heat-transfer data for hot air flowing at superatmospheric pressure at high linear velocities inside a water-cooled tube and report heat-transfer coefficients in terms of equation (3), making no mention of equation (2). The temperature differences were so large, however, that it would have made little difference whether equation (3) or equation (2) was used; hence these data throw no light on the relative advantages of h_s and h_e at moderate and low temperature differences.

From an analytical study of flow of high-velocity streams of air in tubes with heat transfer, it is concluded in reference 11 that h_e is to be preferred, and it is predicted that the recovery factor depends on both the Prandtl number and the Reynolds number based on tube diameter. The corresponding predicted values of r are given in figure 2 of reference 5.

The object of this investigation is (a) to compare the variations with temperature difference of the coefficients of heat transfer defined in terms of mean stream temperature, stagnation temperature, and adiabatic wall temperature, respectively, and (b) to find the effect of variation in Mach number on the coefficient of heat transfer for Mach numbers less than 1. The investigation was limited to heat transfer to air flowing through a smooth brass tube.

This investigation, conducted at the Massachusetts Institute of Technology was sponsored by, and conducted with the financial assistance of, the National Advisory Committee for Aeronautics.

DESCRIPTION OF APPARATUS

In order to simulate the conditions of heat transfer in an aircraft heat exchanger at high altitude a heated tube of small diameter, 0.281 inch, was used for the test section and air at subatmospheric pressure was passed through this tube at velocities corresponding to Mach numbers from 0.1 to 1.

The general arrangement of the apparatus is shown in figure 1. Air at atmospheric pressure and room temperature enters the upstream chamber, where its inlet stagnation temperature is measured. Then it flows in turn through a calibrated metering nozzle, a pipe 127.25 inches long which is insulated with a 2-inch layer of glass wool, a heated pipe 15.25 inches long of the same diameter as the preceding insulated length, a baffled mixing chamber in which the outlet mean stagnation temperature is measured, a throttle valve, and a two-stage steam ejector. Wall temperatures and stream pressures are measured at the points indicated in figure 1.

The heated section and its steam jackets are shown in figure 2. Tube-wall temperatures are measured at four points in the heated section. Noncondensable gas is vented from the inner jacket continuously at two points. By connecting the vents through valves to the ejector the pressure in the steam jackets could be made subatmospheric. Condensate is withdrawn from the inner jacket at a constant rate by adjusting a valve in the condensate line so as to maintain a constant level of condensate in a sump consisting of a gage glass.

In order to reduce the heat transfer from the steam jacket to the mixing chamber the diaphragm separating the two was made of Bakelite 2 inches thick. To reduce heat transfer from the jacket to the unheated pipe, the thickness of the tube wall was reduced from 0.062 to 0.01 inch for a length of 2 inches immediately adjacent to the upstream end of the heated length.

In order to obtain a heat balance, the heat transferred from the tube wall to the air stream was measured on the one hand by the mass rate of flow of air and the change in the mean stagnation temperature of the stream, and the heat transferred from the steam was measured on the other by the mass rate of condensation. The measured condensate was formed in the jacket surrounding the test pipe. To prevent radial transfer of heat from this jacket to the surroundings, an isothermal environment was provided by a second jacket surrounding the first. Steam entered the outer jacket and entered the top of the inner jacket through a trap designed to admit steam but not condensate.

The mixing chamber was insulated with 2 inches of hair felt, which was heated externally by electrical heating coils. Embedded in the insulation were six thermocouples, each with one junction at the outer surface of the wall of the mixing chamber and the other junction halfway between the wall and the outside surface of the insulation. The supply of current to the heating coils was adjusted until the indications of the thermocouples corresponded to negligible heat flow through the insulation.

Wall temperatures along the test pipe were measured with calibrated copper-constantan thermocouples and a portable potentiometer sensitive to 0.2° F. Cold junctions were maintained at 32° F.

TEST PROCEDURE

Constant air rates were obtained with very little control apparatus by keeping the steam ejector wide open. For this condition the pressure on the downstream side of the throttle valve was less than 1 inch of mercury absolute. As this value was well below the sound pressure (the pressure of maximum entropy) for all runs, the inlet and outlet pressures of the test pipe became barometric pressure and the sound pressure, respectively.

The value of the Mach number at exit could be reduced by partly closing the throttle valve in the discharge pipe. The value of the Reynolds number could be altered independently of the Mach number by partly closing a globe valve in the $1\frac{1}{2}$-inch pipe preceding the test pipe.

The data taken for each run are given in table 1. For the adiabatic runs, equilibrium was assumed when the wall temperature remained constant for half an hour. A period of 2 to 3 hours was necessary to attain this condition. For the heat-transfer runs, equilibrium was assumed when the temperature in the mixing chamber remained constant for half an hour after the differential thermocouples in the insulation on the chamber were brought to substantially zero potential by means of the electrical heating coil. A period of 3 to 4 hours was necessary to attain this condition.

After equilibrium was reached, the data for an adiabatic run were taken in about 10 minutes. When equilibrium was reached in a heat-transfer run, values were recorded for the quantities listed in table 1. Then, as the condensate collected, the inlet stagnation temperature, outlet stagnation temperature, and wall temperatures were recorded every 5 to 10 minutes for a half hour. The condensate then was weighed and all other readings were again taken.

RESULTS

Recovery Factors

In order to evaluate recovery factors for an adiabatic run, it is necessary to compute the mean temperature of the stream at various distances from the nozzle. From the perfect gas law ($\rho_m = p/RT_m$), the equation of continuity ($V_m = G/\rho_m$), and the energy balance for adiabatic flow ($T_s - T_m = V_m^2/2g_cJc_p$), the values of T_m at various lengths were computed as the ~~real~~ positive root of the equation

$$\left(\frac{G^2R^2}{2g_cJc_pp^2}\right)T_m^2 + T_m - T_s = 0 \qquad (5)$$

using observed values of pressures and stagnation temperatures from table 1.

To allow for the nonuniformity in velocity distribution across the pipe, the kinetic term should be written

$$\frac{V_m^2}{2g_c\alpha Jc_p}$$

where the dimensionless factor α depends slightly on the Reynolds number. At the nozzle the velocity distribution is uniform and consequently α is 1.0, but as the air flows down the tube the value of α asymptotically approaches the normal value (somewhat less than 1) for fully developed flow. However, the usual procedure of taking α as 1 has been followed.

Figure 3, based on data of run 53a (table 1), shows the results of an adiabatic run made at a mass velocity of 18 pounds per second per square foot of cross section, corresponding to a Reynolds number averaging 35,000. As the air flows through the tube the stagnation temperature remains constant at room temperature, the mean temperature falls off sharply owing to marked acceleration at the downstream end, and the temperature of the adiabatic wall always lies above the mean temperature and below the stagnation temperature.

Recovery factors are calculated by applying equation (4). From figure 3 it is seen that all three temperatures (t_s, t_{aw}, and t_m) differ but little near the nozzle, and the corresponding values of r consequently have little precision there. At the discharge end the values of r are the most precise. The relatively small variation in recovery factor with distance from the nozzle might be attributed to one or more of the following factors: increase in Mach number, gradual change in velocity distribution from that characteristic of a tube, longitudinal conduction of heat through the tube wall, and inward heat flow through the insulation. According to reference 12, increase in Mach number, for flow of air past a flat plate with a laminar boundary layer, should slightly decrease r. Recovery factors for the other adiabatic runs vary even less than for the run shown in figure 3.

The results for all adiabatic runs are summarized to a highly magnified ordinate scale in figure 4. In the jacketed portion of the tube, where the precision is best, the recovery factors range only from 0.875 to 0.905; near the nozzle, where any effect of conduction is negligible, but precision is poor, r ranges from 0.80 to 0.875, with an average value of 0.85.

Heat Balances

From the measured mass velocity G (table 3) and the terminal stagnation temperatures t_{so} and t_{si} (table 1) the heat transfer rate q_a to the air stream was calculated from the equation

$$q_a = wc_p \, (t_{so} - t_{si})$$

using a value of c_p of 0.240. From the observed mass rate of flow of condensate w_c from the inner jacket (table 1) the corresponding heat-transfer rate q_v was calculated from the latent heat corresponding to the saturation pressure ($q_v = \lambda w_c$).

In order to determine any discrepancy in the heat balances the ratio q_a/q_v (table 3) was calculated. Owing primarily to heat losses at the ends of the heated section the ratio q_a/q_v should be less than 1.0. In the runs with steam at the highest pressure, where the heat-transfer rates are highest and the errors in q_a/q_v are consequently lowest, the ratio ranges from 0.93 to 1.04, averaging 0.98. Since the values of q_a are believed to be more dependable than those for q_v, all heat-transfer coefficients are based on q_a.

Calculation of Coefficients of Heat Transfer

The three coefficients of heat transfer, h_m, h_s, and h_e, for the heated section are calculated from the temperatures t_m, t_s, and t_{aw} at entrance and exit of the heated section. The value of t_s at entrance is the temperature of the room from which the air is supplied. The value of t_s at exit is the temperature in the mixing chamber. The values of T_m at entrance and exit are found from equation (5) by substituting for T_s the appropriate value.

The values of t_{aw} at entrance and exit are found by substituting into equation (4) the appropriate values for t_s and t_m and the values of r as found for the same positions along the tube in adiabatic tests. (An alternative method of determining t_{aw} in which t_s was reduced by the difference between t_s and t_w in the corresponding adiabatic test yielded coefficients differing only by 1 percent from those from the first method.)

For each test a coefficient h_m was assumed, and this was used to compute the change in stagnation temperature Δt_s for a short length of the heated section by means of the equation

$$\Delta t_s = \frac{h_m \Delta A(t_w - t_m)}{wc_p}$$

This increment in t_s determined the value of t_s at the new position along the pipe. This value, together with equation (5), determines the new value of t_m. The new value of t_w is found by graphical interpolation between the six measured values.

With the same assumed value of h_m computations were made for successive intervals of length until the final value of t_s was found. This temperature was compared with the temperature of the mixing chamber. If the calculated value of the final stagnation was in excess of the mixing-chamber temperature, a lower value of h_m was assumed and the calculation repeated. When the calculated and measured values are brought to coincidence, the assumed value of h_m is considered to be the experimental value.

The same method was used for determining h_s and h_e, the corresponding equations being, respectively,

$$\Delta t_s = \frac{h_s \Delta A(t_w - t_s)}{wc_p}$$

and

$$\Delta t_s = \frac{h_e \Delta A(t_w - t_{aw})}{wc_p}$$

Figure 5 shows the calculated variation in t_m, t_s, and t_{aw} and the measured value of t_w for run 73.

Comparison of Coefficients

The variation of the measured coefficients of heat transfer, h_m, h_s, and h_e with stagnation-temperature difference is shown in figure 6 for a series of seven tests with Mach numbers at entrance and exit of approximately 0.5 and 1, respectively. The mass velocities were substantially the same for all seven runs, varying from 17.4 to 17.9 pounds per second per square foot. The corresponding variation in Reynolds number, based on the viscosity at the mean stream temperature, was from 33,800 to 36,000.

The coefficient h_e shows no detectable variation with temperature difference. The coefficient h_s appears to be nearly constant for temperature differences of 100° F or more, but for smaller temperature differences it increases and reaches infinity when the stagnation-temperature difference is zero. It is less than zero when the wall temperature lies between the stagnation temperature and the adiabatic wall temperature. The coefficient h_m is the least satisfactory of the three. It approaches constancy only at very great temperature differences, and it is zero when the wall temperature coincides with the adiabatic-wall temperature.

Similar results are shown in figure 7 for six runs at lower Reynolds number. The values of the Reynolds number for these runs ranged from 24,000 to 24,900.

As the wall temperature is increased for fixed stream conditions, the coefficients h_e and h_s approach each other in magnitude. The difference between the two depends upon the ratio of the temperature difference causing heat transfer to the temperature interval $(t_s - t_{aw})$. This observation may be formulated, in view of the present data, as follows.

If the wall temperature exceeds the stagnation temperature by an amount which is greater than twice the difference between the stagnation temperature and the mean stream temperature, the value of h_s will be nearly independent of the difference between wall temperature and stagnation temperature.

The wall temperature must exceed the mean stream temperature by an amount many times the difference between the stagnation temperature and the mean stream temperature if h_m is to be even approximately independent of the difference between wall temperature and mean stream temperature. Compared with h_e and h_s, the coefficient h_m will seldom prove serviceable.

Variation of Coefficient with Reynolds Number and Mach Number

The variation of the preferred Stanton number, h_e/c_pG, with Reynolds number is shown in figure 8. The results of the present measurements at average Mach numbers ranging from 0.1 to 0.2 are satisfactorily represented by the relation

$$\frac{h_e}{c_pG} = 0.025 \left(\frac{DG}{\mu_m}\right)^{-0.2} \tag{6}$$

As shown by figure 8, this expression is further substantiated for lower Mach numbers by the data of reference 13, which had good heat balances. Published data which do not include heat balances (references 14, 15, and 16) yield a spread of points corresponding to constants in equation (6), of 0.024 to 0.032, with an average of about 0.027. All the published data were obtained with less calming length preceding the heated section than in the present apparatus.

The present data for Mach numbers ranging from 0.32 to 0.51 lie only 4 percent below the dotted curve representing equation (6), and those for Mach numbers from 0.43 to 1 lie only 8 percent below.

References 9 and 10 report tests on the cooling of hot air flowing at high velocities in tubes at pressures greater than atmospheric. In reference 9 the diameter and length of the tube were 0.551 inch and 56.4 inch, respectively, and in reference 10 the corresponding dimensions were 0.985 inch and 99.3 inches. Since the temperature differences were larger than any used in the present study, these data throw no light on whether h_e or h_s should be used. In these same tests the Mach number at the exit was 1, the densities and diameters were larger than in the present investigation, and the Reynolds numbers consequently were greater. Since figures 6 and 7 show that h_e is preferable to h_s and h_m, the data of references 9 and 10 have been recalculated to determine values of h_e corresponding to a recovery factor r of 0.88. These results are shown in figure 8 along with the results of the present measurements. They lie 5 to 14 percent below those obtained from equation (6).

All the data of figure 8 can be represented within ±7 percent by the relation

$$\frac{h_e}{c_p G} = 0.033 \left(\frac{DG}{\mu_m}\right)^{-0.23} \tag{7}$$

and cover the following ranges of variables:

Diameters, inch	0.28 to 0.99
Absolute pressures, atmospheres	0.2 to 3
Temperature differences, °F	10 to 400
Mach numbers	0 to 1
Reynolds numbers	10,000 to 400,000

Since viscosity enters equation (7) only to the 0.23 power, the constant would be changed by less than 2 percent if viscosity had been based on t_f, defined as $\frac{t_w + t_{aw}}{2}$, instead of t_m.

CONCLUSIONS

From an investigation to compare the variation of the coefficients of heat transfer with temperature difference and to find the effect of variation in Mach number on these coefficients for Mach numbers less than 1, the following conclusions are drawn:

1. The only published heat-transfer coefficients involving high-velocity air streams were measured with such large temperature differences as to throw no light on how the coefficient of heat transfer should be defined.

2. For steady mass flow of air at high but subsonic velocity through a heated tube, the coefficient of heat transfer h_e, based on the difference between the temperature of the heated wall and the adiabatic wall temperature, is independent of this temperature difference. The analogous coefficients h_s and h_m, based on stagnation and mean stream

temperatures, respectively, are not independent of temperature difference, although for temperature differences several times the difference between t_s and t_m the coefficient h_s is nearly independent of the temperature difference.

3. For a given Reynolds number variation in average Mach number from 0.2 to 0.75 has but little effect on the preferred type of coefficient of heat transfer h_e.

4. The present measurements of h_e for Mach numbers ranging from 0.1 to 1, previously published coefficients for incompressible turbulent flow of air, and previously published data for cooling air at high but subsonic velocities, are correlated within ±7 percent by the relation

$$\frac{h_e}{c_p G} = 0.033 \left(\frac{DG}{\mu_m}\right)^{-0.23}$$

These data cover the following range of variables: diameter from 0.28 to 0.99 inch, pressure from 0.2 to 3 atmospheres absolute, temperature difference from 10° to 400° F, Mach number from 0 to 1, and Reynolds number from 10,000 to 400,000. Since viscosity enters the equation only to the 0.23 power, the constant would be changed by less than 2 percent if viscosity had been evaluated at t_f, defined as $\frac{t_w + t_{aw}}{2}$, instead of at t_m.

5. The measured recovery factors for the wall of a pipe are substantially independent of Mach number in the range 0.2 to 1, and average 0.88, which is substantially the same as published values for the flow of air parallel to flat plates and wires. No published data are available for the wall of a pipe.

Massachusetts Institute of Technology,
Cambridge, Mass., February 22, 1945.

APPENDIX

SYMBOLS

A area of heat-transfer surface, sq ft

c_p specific heat at constant pressure, Btu/(lb)(°F)

D inside diameter, ft

d differential operator

G mass velocity, lb/(sec)(sq ft)

g_c conversion factor, mass times acceleration divided by force, 32.2 lb-ft/(sec²)(lb force) or 4.17 × 10⁸ lb-ft/(hr²)(lb force)

h coefficient of heat transfer, Btu/(hr)(sq ft)(°F)

$$h_e = dq/dA\ (t_w - t_{aw})$$

$$h_m = dq/dA\ (t_w - t_m)$$

$$h_s = dq/dA\ (t_w - t_s)$$

J conversion factor, ft-lb/Btu

k thermal conductivity, Btu/(hr)(ft²)(°F/ft)

N_M Mach number, V/Va

N_R Reynolds number, $DG/\mu_m g_c = 4w/\pi D\mu_m g_c$

P pressure, in units specified in tables

P_v pressure of condensing vapor in jacket, in units specified in tables

p absolute pressure, lb force/sq ft

q_a heat-transfer rate to air stream, Btu/hr

q_v heat-transfer rate from condensing vapor, Btu/hr

R gas constant, $p/\rho T$, ft-lb/(lb)(°F absolute)

r recovery factor, dimensionless,

$$r = \frac{t_{aw} - t_m}{t_s - t_m} = \frac{t_{aw} - t_m}{V_m^2/2g_c J c_p}$$

T_{aw} absolute temperature of adiabatic wall, °F absolute

T_m absolute temperature of mean stream, °F absolute

T_s absolute stagnation temperature, °F absolute

T_w absolute temperature of inner surface of heated wall, °F absolute

t_{aw} temperature of adiabatic wall, °F

t_m temperature of mean stream, °F

t_{si} stagnation temperature at inlet, °F

t_{so} stagnation temperature at outlet, °F

t_v temperature of condensing vapor in jacket, °F

t_w temperature of inner surface of heated wall, °F

V_a acoustic velocity, ft/sec or ft/hr

V_m mean velocity, volumetric rate per unit cross section, ft/sec or ft/hr

v specific volume of fluid, cu ft/lb

w mass rate of flow, lb/sec or lb/hr

w_c mass rate of flow of condensate from inner jacket, lb/hr

x distance downstream from nozzle, in.

α velocity-distribution factor, dimensionless, taken as 1.0

ΔA increment in heat-transfer surface, used only in stepwise calculations based on equations (2), (3), and (4), sq ft

ΔP_N pressure drop across calibrated nozzle, cm of water

Δt_s increment in stagnation temperature, used only in stepwise calculations based on equations (2), (3), and (4), $^{\circ}F$

λ latent heat of condensation at saturation pressure, Btu/lb

μ_m absolute viscosity of air at average t_m, (lb force) (sec)/ft^2 or (lb force) (hr)/ft^2 (values based on data from reference 17)

ρ density of air calculated from perfect gas law, lb/cu ft

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TABLE I

ORIGINAL DATA FOR ADIABATIC AND HEAT-TRANSFER RUNS

Run No.	Barometer Inches Mercury Absolute	ΔP_N Cm. Water Difference	P cm of Mercury, Vacuum: Approach Pipe	x, inches: 70	124.5	142.4	P_V Cm. Mercury Gauge
52a	30.097	38.85	-	20.75	41.90	60.95	-
53a	29.646	38.00	-	20.35	41.15	60.00	-
54a	29.977	37.95	-	21.15	42.05	61.10	-
55	30.020	6.10	-	3.25	5.57	6.55	+0.20
56	30.081	17.85	-	9.75	16.58	19.80	+0.20
59a	29.919	32.20	-	16.85	31.60	38.45	-
59	29.792	29.20	-	17.65	31.15	38.75	+0.30
60	29.790	11.45	-	6.30	10.60	12.55	+0.30
61	29.790	35.25	-	21.00	39.95	60.00	+0.30
62	30.156	35.40	-	21.00	39.70	60.65	+0.20
63	29.838	35.60	-	21.75	41.40	60.60	-58.80
64	29.860	35.60	-	21.55	41.05	60.57	-46.50
65	30.070	36.80	-	20.80	40.55	60.75	-27.15
69a	30.364	26.35	21.25	36.70	52.35	66.25	-
70	30.314	25.00	22.70	37.35	51.85	66.25	-51.80
71	30.115	36.70	-	22.25	42.70	61.70	-70.45
72	30.146	36.70	-	22.05	42.45	61.70	-68.90
73	29.998	24.20	22.90	38.20	52.70	66.15	-67.25
74	30.124	24.80	22.40	37.95	52.80	66.30	-70.50
75	30.138	24.30	22.85	38.00	52.40	66.10	-56.60
76	29.903	24.40	21.60	36.30	50.35	65.10	+ 0.20
77	29.933	18.35	-	10.10	17.05	20.50	+ 0.60
78	30.165	24.20	22.35	37.10	52.70	65.80	-38.00
79a	29.881	38.10	-	20.10	41.05	56.40	-
79	30.376	37.65	-	20.00	39.40	56.55	+ 0.40
80	30.264	37.50	-	19.90	39.45	56.78	+ 0.20
81a	29.596	35.05	-	18.35	35.75	44.70	-
81	29.680	33.35	-	17.45	33.25	43.10	+ 0.30
82	29.842	33.90	-	17.85	34.00	44.40	+ 0.30

TABLE I
(CONTINUED)

Run No.	Wall temperatures, t_w, degrees C.: x=25"	x=95"	x=125"	x=126.25"	x=127.25"	x=132.25"
52a	26.15	25.9	25.40	25.30	25.15	24.75
53a	27.35	27.2	26.65	26.50	26.4	26.0
54a	23.90	23.55	23.15	23.0	22.9	22.35
55	26.9	27.2	32.2	48.3	87.2	100.0
56	27.65	27.85	31.2	44.1	86.4	99.7
59a	26.75	26.55	26.4	26.35	26.35	26.25
59	25.9	25.9	28.9	41.5	82.4	99.5
60	26.9	27.15	31.7	45.8	85.5	99.7
61	26.4	26.5	28.4	40.9	81.7	99.5
62	24.4	24.2	26.7	39.0	81.8	99.7
63	26.9	26.7	27.15	32.2	45.8	62.5
64	26.9	26.7	28.2	36.0	59.1	74.6
65	27.0	27.25	29.2	38.75	69.8	86.0
69a	25.65	25.35	24.9	24.85	24.8	24.4
70	24.75	24.6	26.9	38.75	55.7	71.0
71	26.5	26.4	26.3	29.4	33.4	40.7
72	25.9	25.65	25.9	31.7	38.9	47.0
73	28.3	28.2	29.4	35.7	44.2	48.7
74	26.75	26.7	26.9	29.9	33.4	39.5
75	26.4	26.4	28.35	41.0	57.4	66.5
76	25.9	25.9	30.6	53.4	79.6	100.0
77	26.9	26.9	32.2	54.0	83.6	100.2
78	26.9	26.9	30.0	46.3	68.5	81.3
79a	28.5	28.25	27.75	27.7	27.6	27.2
79	26.15	26.15	29.2	46.6	83.5	100.5
80	27.6	27.15	30.2	48.2	83.2	100.0
81a	24.8	24.55	24.4	24.35	24.3	24.15
81	25.25	25.25	28.35	49.0	82.4	99.5
82	24.75	24.75	28.2	46.5	82.7	100.0

TABLE I
(CONCLUDED)

Run No	t_w, degrees C x=137.25"	t_w, degrees C x=141.5"	t_{si} °C.	t_{so} °C.	t_v °C.	w_c lb./hr.
52a	24.15	22.85	27.0	27.0	-	-
53a	25.25	24.0	28.1	28.1	-	-
54a	21.6	20.35	24.6	24.6	-	-
55	100.0	99.7	26.95	68.7	100.2	0.222
56	99.7	99.5	27.8	65.9	100.2	0.347
59a	26.15	25.9	27.1	27.2	-	-
59	99.5	99.3	26.6	63.3	100.0	0.419
60	99.7	99.5	27.3	67.3	100.0	0.287
61	99.5	99.0	27.2	62.4	100.0	0.440
62	99.5	99.3	24.8	61.3	100.2	0.436
63	62.5	62.4	27.5	45.2	63.0	0.196
64	74.6	74.4	27.4	51.2	75.0	0.267
65	86.0	86.0	28.0	57.0	88.0	0.343
69a	23.6	22.5	26.1	26.2	-	-
70	71.5	71.5	25.5	49.6	72.0	0.203
71	40.7	40.7	27.1	34.6	42.0	0.0728
72	47.0	47.0	26.4	37.4	47.5	0.115
73	48.7	48.7	29.2	40.6	49.6	0.0927
74	39.5	39.5	27.4	34.6	41.5	0.0265
75	66.5	66.5	27.1	48.3	68.0	0.182
76	100.0	100.0	26.6	64.7	100.1	0.309
77	100.2	100.2	27.1	66.0	100.3	0.352
78	81.3	81.3	27.9	56.3	82.5	0.213
79a	26.7	26.0	29.1	29.1	-	-
79	100.5	100.5	26.8	62.9	100.7	0.450
80	100.0	100.0	28.3	63.3	100.4	0.441
81a	23.9	22.6	25.3	25.4	-	-
81	99.7	99.7	25.5	63.1	100.0	0.430
82	100.0	100.0	25.1	63.1	100.0	0.454

TABLE 2

CALCULATED RESULTS FOR ADIABATIC RUNS

Run No.	T_m, Mean Temperature Deg. R.				Mach Numbers			
	x inches				x inches			
	0	70	124.5	142.4	0	70	124.5	142.4
59a	563.3	333.5	528.7	524.2	0.205	0.255	0.338	0.398
81a	532.5	529.5	521.4	512.5	0.216	0.275	0.393	0.496
79a	537.9	535.6	522.0	482.6	0.224	0.293	0.464	0.797
53a	537.2	533.0	519.9	455.1	0.225	0.296	0.470	0.982
54a	530.9	526.3	513.3	447.1	0.224	0.297	0.473	1.00
52a	535.0	531.2	517.5	452.6	0.226	0.297	0.473	0.987
69a	534.0	530.2	517.9	451.3	0.216	0.288	0.464	0.987

TABLE 2 (CONCLUDED)

Run No.	G lb /sec ft²	Recovery Factors, $(t_{aw}-t_m)/(t_s-t_m)$						
		x inches						
		20	40	60	80	100	120	140
59a	16.69	0.88	0.88	0.88	0.88	0.88	0.89	0.88
81a	17.37	0.85	0.85	0.85	0.86	0.87	0.89	0.88
79a	18.08	0.85	0.85	0.85	0.84	0.86	0.88	0.89
53a	18.02	0.80	0.82	0.84	0.85	0.87	0.88	0.89
54a	18.22	0.81	0.83	0.83	0.84	0.85	0.87	0.88
52a	18.39	0.78	0.79	0.80	0.81	0.82	0.85	0.90
69a	12.92	0.87	0.87	0.86	0.86	0.86	0.90	0.91

TABLE 3

CALCULATED RESULTS FOR HEAT TRANSFER RUNS

Run No	G lb/sec ft²	q_a Btu/hr.	q_v Btu/hr.	q_a/q_v	Mach Number x, inches 124.5	Mach Number x, inches 142.4	$(t_w-t_s)_{ave}$ °F.
55	7.20	201	216	0.93	0.092	0.100	88.2
60	9.88	265	278	0.95	0.139	0.150	89.0
56	12.40	316	337	0.94	0.188	0.211	90.5
77	12.58	328	341	0.96	0.192	0.218	92.0
59	15.86	392	407	0.96	0.320	0.406	94.1
81	16.92	426	417	1.02	0.359	0.494	95.9
82	17.17	437	440	0.99	0.367	0.511	96.3
79	18.19	440	437	1.01	0.430	0.799	96.5
80	18.08	424	428	0.99	0.430	0.796	94.0
71	17.85	89.6	75.5	1.19	0.469	1.010	16.8
72	17.88	132	118	1.12	0.464	1.008	25.0
63	17.52	208	199	1.04	0.455	0.990	44.6
64	17.53	280	267	1.05	0.450	0.993	59.5
65	17.85	347	339	1.02	0.445	0.996	74.5
61	17.42	411	425	0.97	0.435	0.981	94.2
62	17.62	430	423	1.02	0.425	0.975	98.0
74	12.31	58.4	27.4	2.13	0.462	1.009	14.1
73	12.08	92.1	95.0	0.97	0.456	1.010	22.7
75	12.15	173	183	0.95	0.447	0.998	48.3
70	12.43	201	203	0.99	0.431	0.992	57.5
78	12.34	235	211	1.11	0.460	0.991	67.0
76	12.29	311	300	1.04	0.427	0.997	93.6

TABLE 3

(CONCLUDED)

Run No.	h_s Btu/(hr)(sq ft)(deg F)	h_e Btu/(hr)(sq ft)(deg F)	h_m Btu/(hr)(sq ft)(deg F)	h_s/c_pG	h_e/c_pG	h_m/c_pG	N_R x=134"
55	23.3	23.3	23.3	0.00375	0.00375	0.00375	12,800
60	30.8	30.8	30.7	0.00361	0.00361	0.00360	17,600
56	36.2	36.2	35.0	0.00338	0.00338	0.00327	22,100
77	36.4	36.4	36.0	0.00336	0.00336	0.00331	22,500
59	43.3	42.5	38.1	0.00316	0.00310	0.00278	29,000
81	46.1	44.5	38.7	0.00316	0.00304	0.00264	31,200
82	47.0	45.3	38.6	0.00317	0.00306	0.00260	31,700
79	47.3	44.8	33.0	0.00301	0.00285	0.00210	34,700
80	47.0	44.1	32.1	0.00300	0.00284	0.00206	34,400
71	58.0	43.3	13.3	0.00376	0.00281	0.00087	36,000
72	56.0	44.6	16.9	0.00362	0.00288	0.00110	36,000
63	51.1	43.8	21.6	0.00338	0.00289	0.00143	34,700
64	49.0	44.3	24.0	0.00323	0.00292	0.00158	34,600
65	49.0	44.5	26.8	0.00318	0.00289	0.00174	35,000
61	45.9	43.0	28.1	0.00304	0.00286	0.00187	33,800
62	46.6	43.6	28.8	0.00306	0.00287	0.00189	34,400
74	45.5	33.6	9.25	0.00427	0.00316	0.00087	24,900
73	41.7	34.2	11.6	0.00400	0.00328	0.00111	24,300
75	36.8	32.9	16.8	0.00351	0.00314	0.00160	24,100
70	37.2	34.0	18.3	0.00346	0.00316	0.00170	24,600
78	36.3	33.7	19.3	0.00340	0.00316	0.00181	24,200
76	34.7	33.0	21.1	0.00327	0.00311	0.00198	24,000

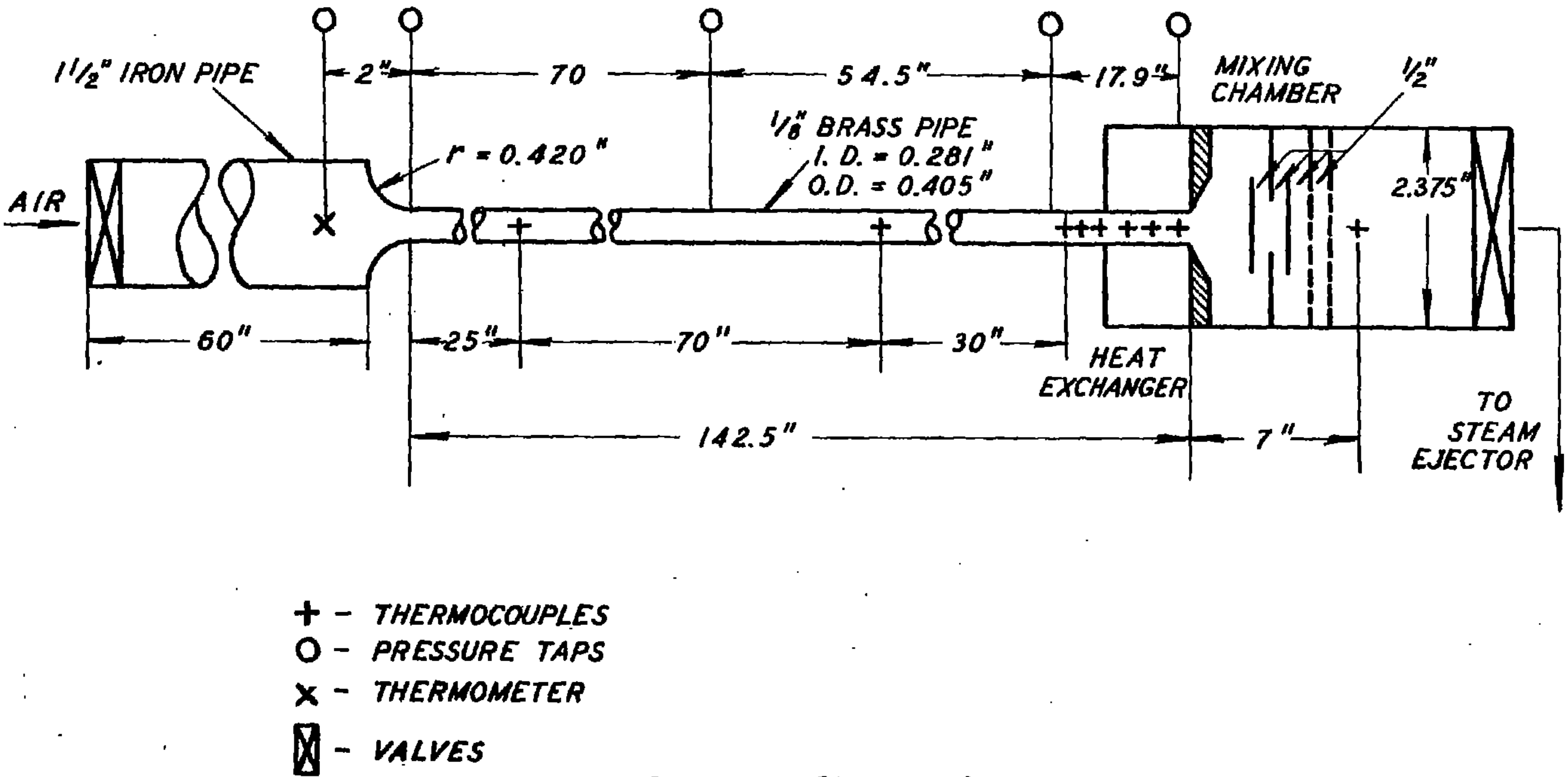

Figure 1.- Diagram of apparatus.

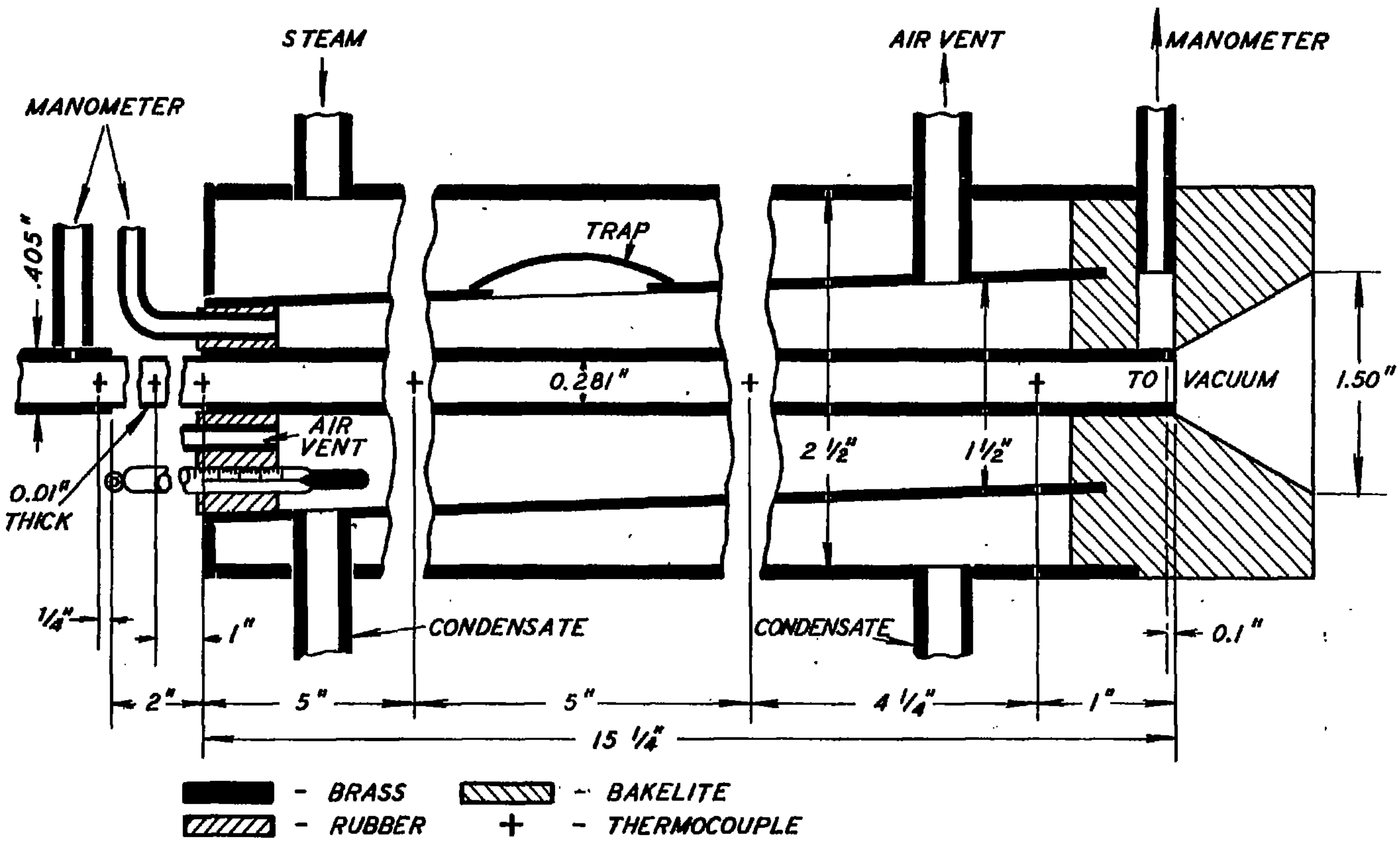

Figure 2.- Longitudinal section of heated length.

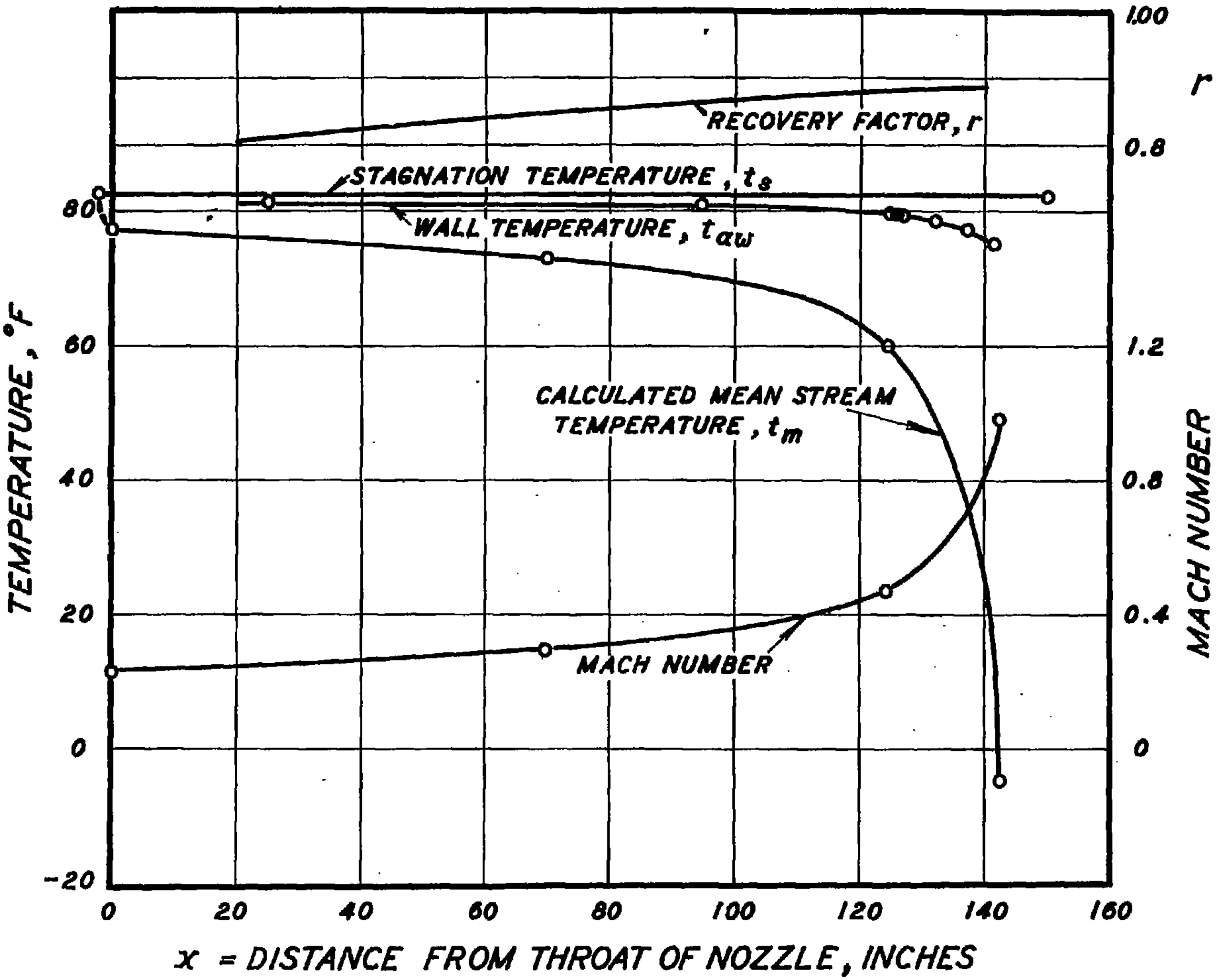

Figure 3.- Temperatures and recovery factors r for adiabatic run 53a, with Reynolds number of 35,000. The Mach number increases from .22 at the entrance to .98 at the exit.

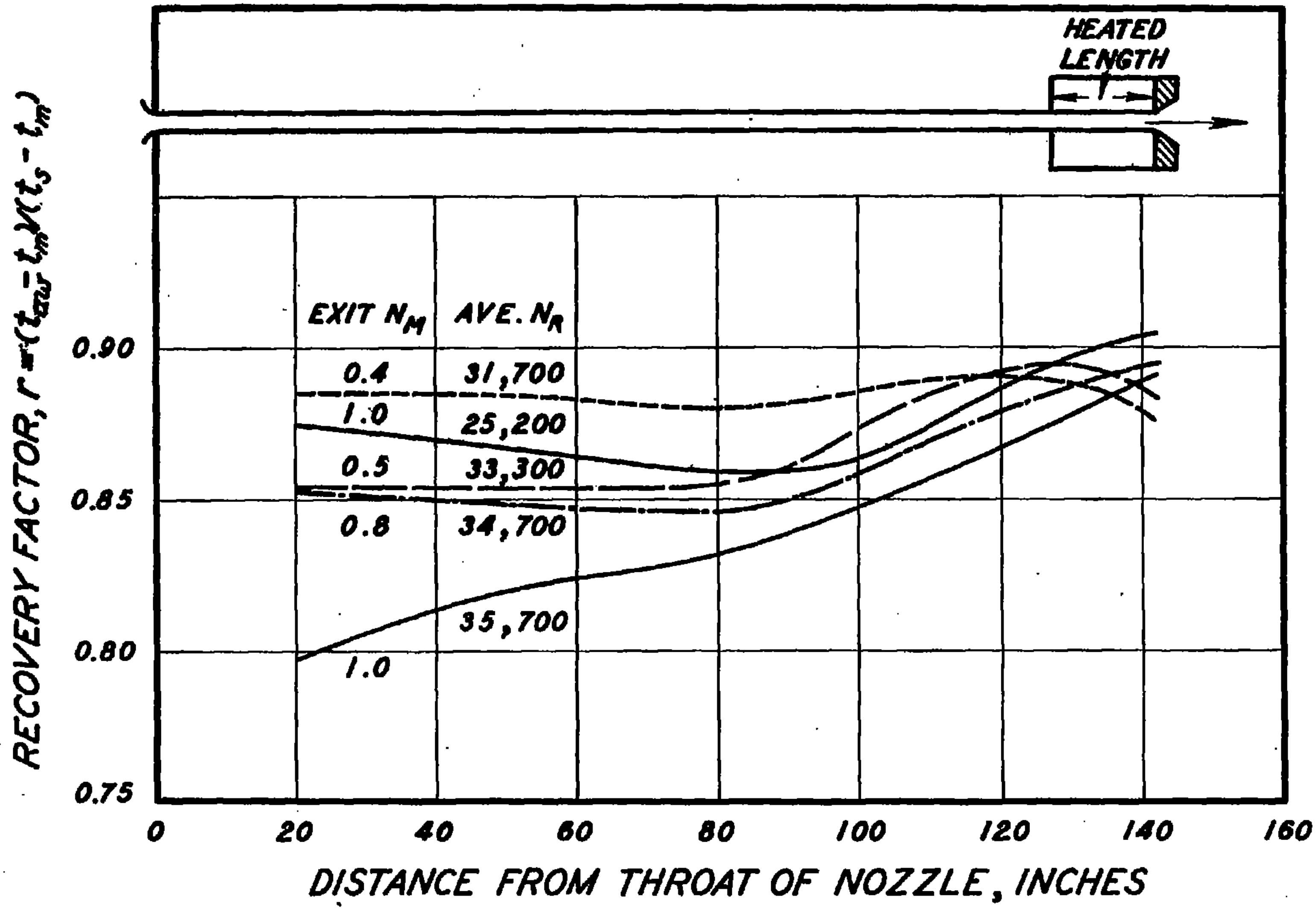

Figure 4.- Faired curves of recovery factors for all adiabatic runs.

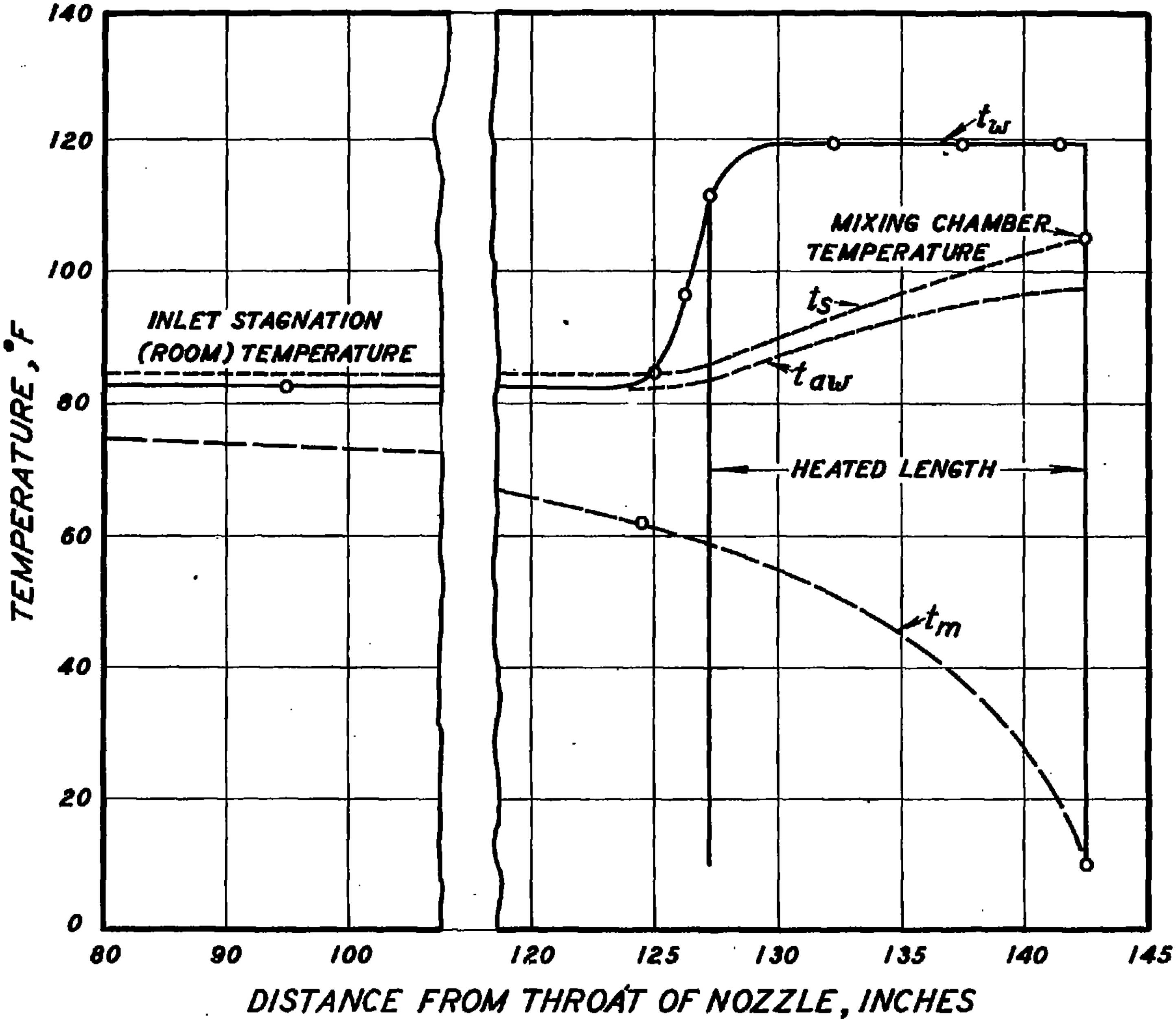

Figure 5.- Temperatures for heat-transfer run 73, with Reynolds number of 24,300. Mach number increases from .46 at the entrance to the heated length to 1.0 at the exit.

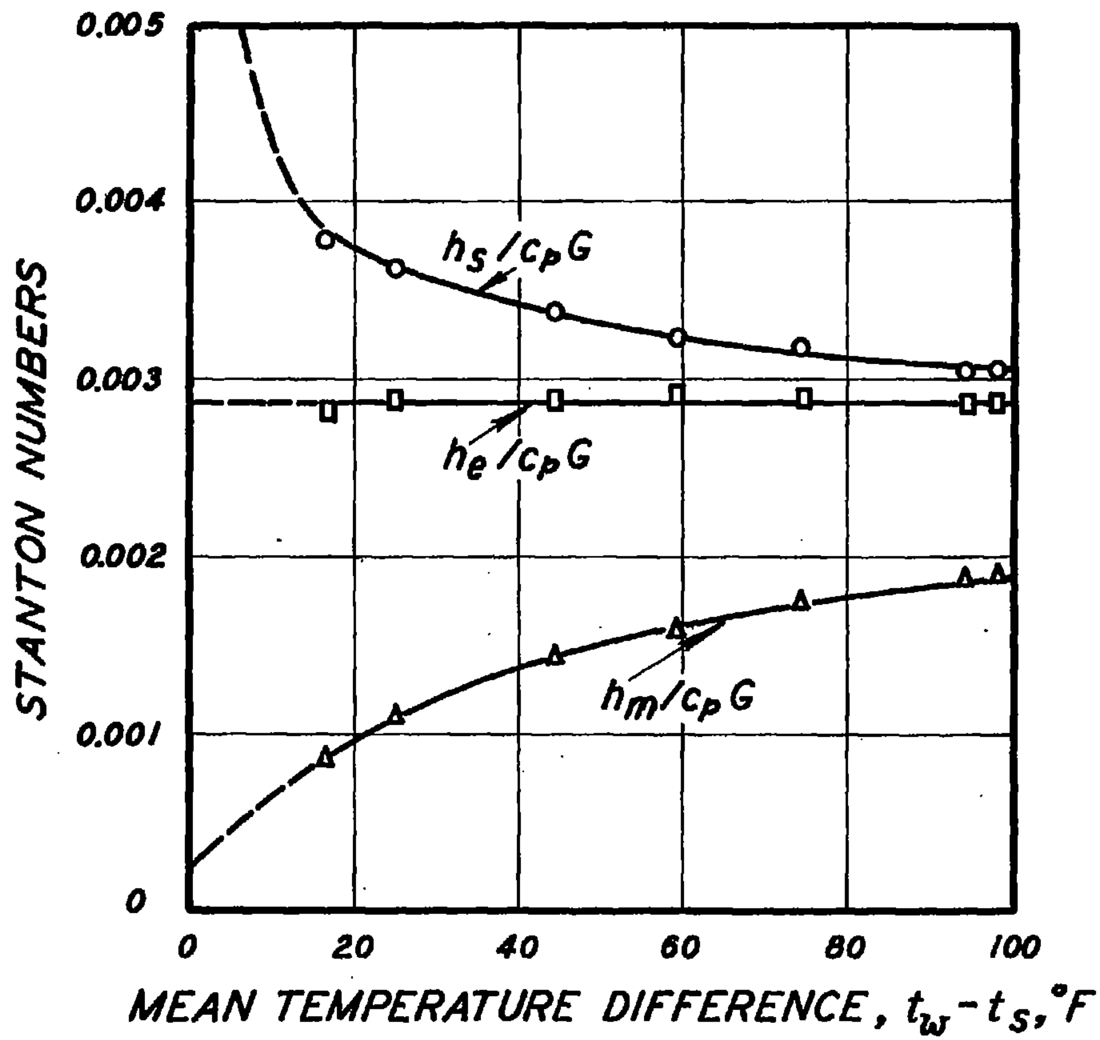

Figure 6.- Effect of temperature difference on the three Stanton numbers for runs at substantially constant Reynolds number (33,800 to 36,000). The Mach number increases from .5 at the entrance to the heated length to 1 at the exit.

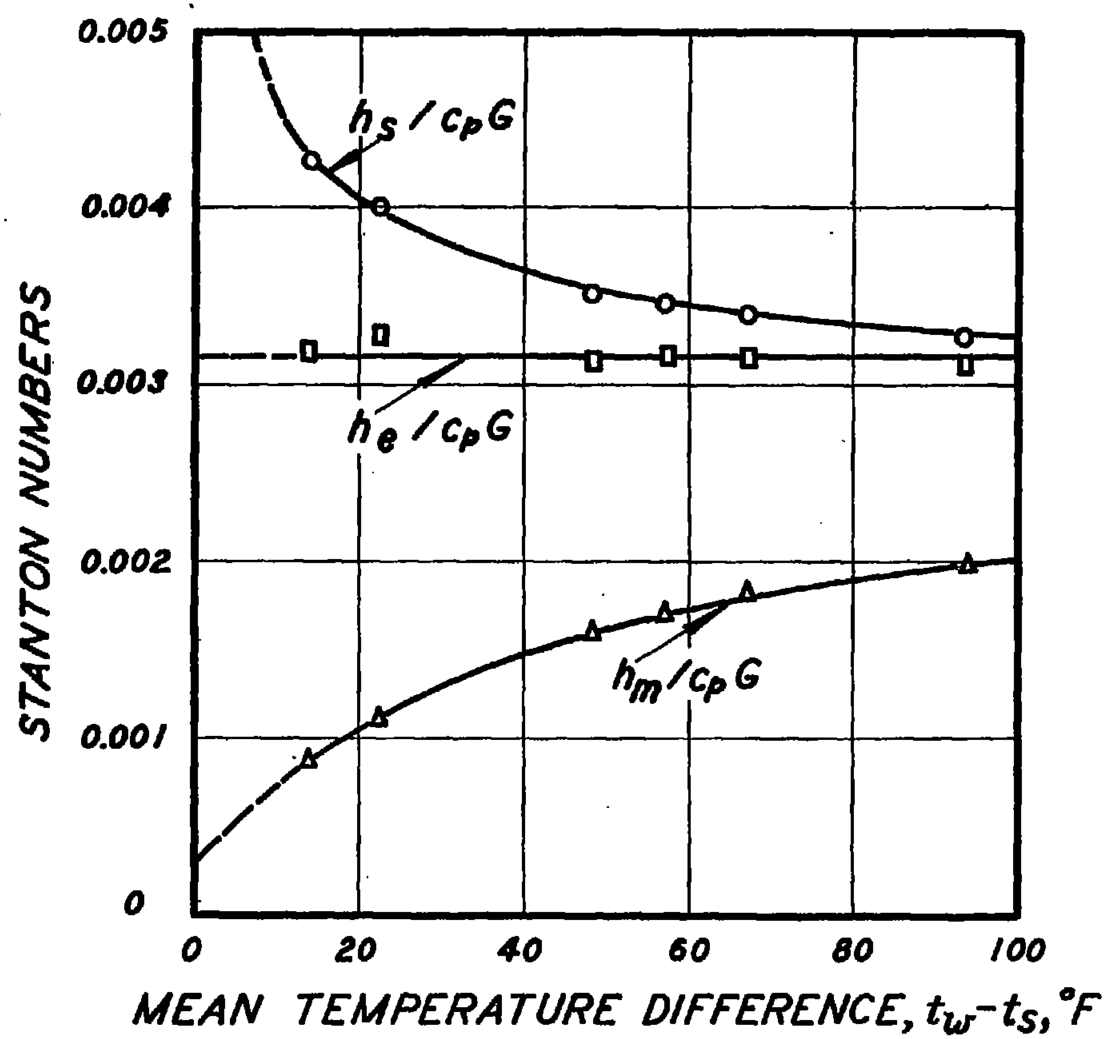

Figure 7.- Effect of temperature difference on the three Stanton numbers for runs at substantially constant Reynolds number (24,000 to 24,900). The Mach number increases from .5 at the entrance to the heated length to 1 at the exit.

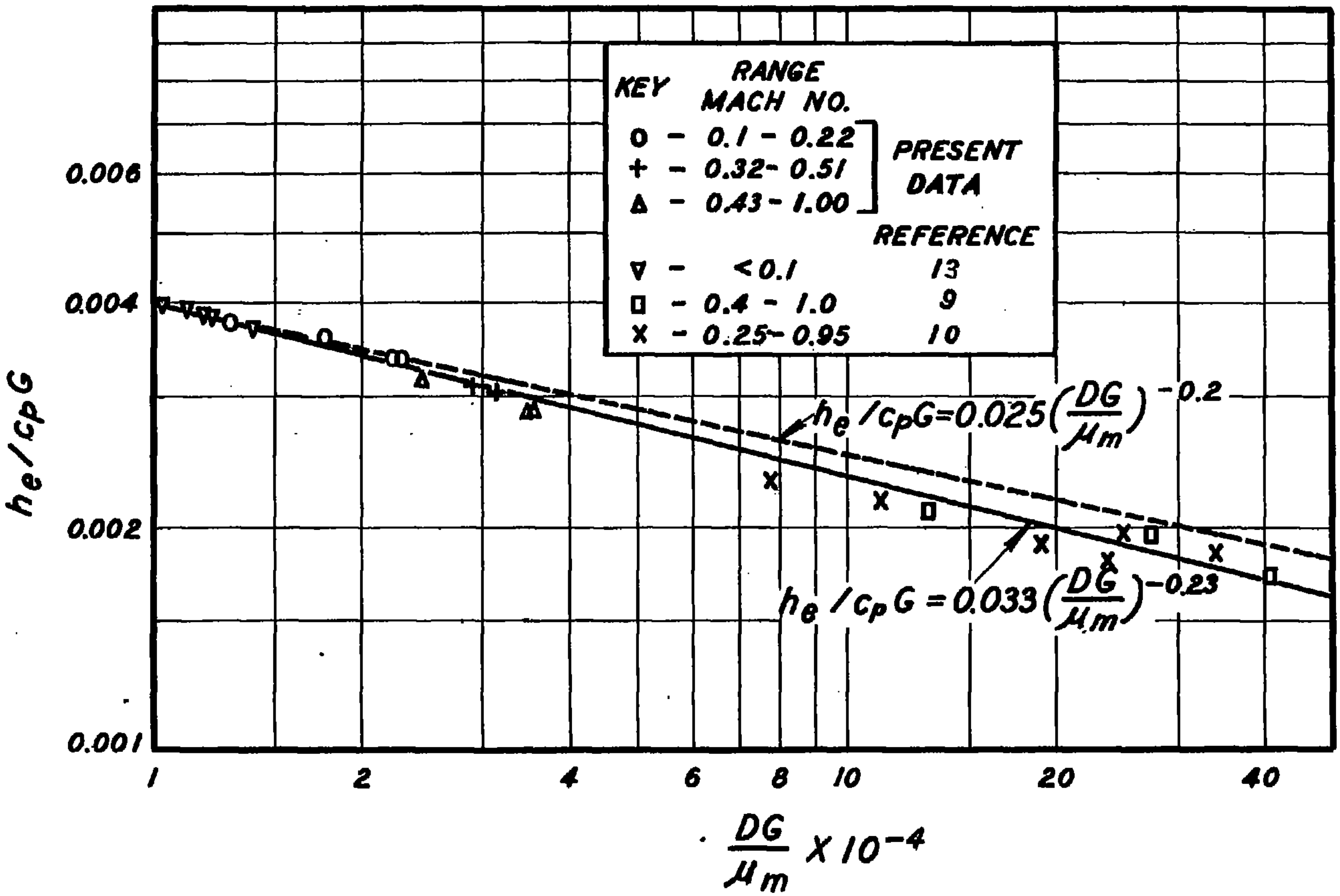

Figure 8.- Effect of Reynolds number on preferred type of Stanton number, at various Mach numbers ranging from 0 to 1.0.